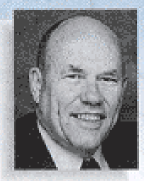
Shop testing - Is it worth it?



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This article discusses the Shop Testing of Equipment that an Original Equipment Manufacturer (OEM) does before a machine is shipped. I have been doing Mechanical and Performance shop testing for about 35 years, so I can offer some perspective. Are there benefits? In my experience, shop testing, as applied, has been worth it, without exception. All critical, unspared equipment is mechanically tested - basically a 4 hour run of the spare and contract rotors, each separately fitted into the casings, and meeting certain criteria, including vibration levels, balance verification, seal leakage, and bearing performance.

I have included Table 1 to guide you in certain areas, outlined by the American Petroleum Institute (API) Standards, which have been broadened in scope to address Petroleum, Chemical, and Gas Industry Services. In some cases, these standards have also been adopted by the International Standards Organization (ISO). In this article, only turbomachinery equipment will be discussed; that is, no reciprocating compressors or engines. The basic format will be taken from API's approach, as I spent over 22 years in the preparation of those standards and commissioned three large machine trains in 1996.

Some comments are appropriate on the items in the Standards that need addressing, and some guidance on items thought to be a waste of money or "free to be waived." Furthermore, it seems many people do not understand where the Owner/Operator of the equipment must make a commitment...these are bulleted paragraphs (•) in the Standards, and a position is necessary on the part of the owner (USER) or his representative Contractor. Many people think that by purchasing, for instance, a steam turbine to API 612, Special-Purpose Steam Turbines, that all the requirements are spelled out. This is NOT SO! While there are fewer bulleted paragraphs than in previous editions of the standards, there are still many remaining.

Furthermore, because the API Standards are written by many companies that often cannot agree on the "added requirements," the bulleted paragraphs give the owner an opportunity to express his desires. Finally, with a few exceptions, these standards are purchasing standards only and do not set forth conditions for Full Speed, Full Load conditions in an operating environment.

Principles I have learned

• During the last 22 years, I have learned some Cost & Reliability Analysis principles that enabled us to operate machines for 15 years at 99% availability and to reduce our machinery costs from 22% of the Maintenance Budget to 16% over a 10 year span. I call these principles the "Law of Fours."

- Production Outage Costs will exceed Mechanical Repair Costs by 4:1
- Conversions in the field exceed those in the OEM shop by 4:1
- Pump Costs will exceed any other mechanical category by 4:1
- The "worst" pump group repairs will exceed "new" replacement by 4:1
- Good Turbo maintenance is \$2-\$4/hp/yr; better than "next" by 1:4

Equipment Type	API Std./RP	Vibration Limit-Test	Unbalance Resp.	Other Requirements
Pumps	610 - 8th, August '95	$(8000/N)^{0.5}$ mil pp, 0.12 ips rms caps	4W/N oz · in (or) ISO 0.665	Vert pump - $(10{,}000/N)^{0.5}$ mil pp, 0.20 ips rms seismic
Steam Turbines (Special)	612 - 4th, June '95	$(12{,}000/N)^{0.5}$ mil pp shaft relative.	4W/N oz · in (or) ISO 0.665 ¶ 2.8.3 Shop Unbalance Response	4 hour running test w/ trip and all ADRE type data - Bode, polar, eccentricity, cascades, w/o Electrical Run Out.
Gears (Special)	613 - 4th, June '95	On the casing: 0.15 ips pk (10 Hz - 2.5 kHz); 4 g pk (2.5kHz -10 kHz)	4W/N oz · in (or) ISO 0.665 mm/s.	Shaft Vibration during shop test w/proximity probes, not to exceed (12,000/N) 0.5 mil pp
Lube Consoles	614 - 3rd, June '95 (4th due)	N/A	N/A	Oil Filtration 10μm. Steam Turbine Main Pump w/ Motor stand-by. Install 2nd Aux. Start Switch @ Main Pump Discharge
Centrifugal Compressor	617 - 6th, February '95	$(12{,}000/N)^{0.5}$ mil pp shaft relative	4W/N oz · in (or) ISO 0.665 mm/s. (Ditto Turbine)	4 hour mechanical run w/performance testing on new designs or multi-section SM per fig. 8, Section 4.3.4
Screw Compressor	619 - 3rd, June '97	$(16{,}000/N)^{0.5}$ mil pp shaft relative	4W/N oz · in (or) ISO 0.665 mm/s.	Silencers- important. Alloy coating w/steel
Fans	673 - 1st '82 being rev'd.	0.1 ips pk on the bearing.- seismic	ISO 2.5 mm/s	Speed was limited to 1200 rpm in 1st draft
Motors, Induction	541 - 3rd April '95	$(12{,}000/N)^{0.5}$	4W/N oz · in	250 + hp

Table 1.

• Not only is a test of both the "spare rotor" and the "contract rotor" reassuring; but I have learned that "often" they do not fit in the same "contract" casing!

• Buying the contract rotor, spare rotor, and "expected consumable" parts - bearings, seals, couplings, and their specific spares at the time the purchase order is signed, and before testing is completed - generates a discount to both the USER/OWNER and OEM. All parts are then manufactured at the same time with the same set up and will be on hand at startup commissioning. Shop Training, at the time of testing milestones, can "bring operating people" in ownership early, and can guarantee completed, reviewed, and corrected manuals at the time of predelivery training.

• Delays of 6-9 months are not uncommon to assure that, for example, compound (multi-section) compressors meet performance and that surge, stonewall, or "no flow" does not exist in the field. The margin to surge in the shop (stability range) can often be "short" of what is seen in the field due to actual field "impedances." This justifies the need for surge and anti-surge testing during commissioning, as well as in the shop. One should agree on whether Reynolds number correction will be allowed during surge and performance testing, before the testing is done. There is a corollary to this on rotor balance and runout checks that says the "acceptable limits" should be agreed upon before the measurements are made!

• It is better to agree on the SAFE "Margins of Separations"(resonances versus operating speed ranges) than to waste much time on confirming the OEM's rotor dynamics programs. While the later standards concentrate on percentage of bearing and seal clearances more than "Amplification Factor" or Damping Factor/Ratio, I have felt more confident when the resonances have been "taken out of play"... sort of like the Professional Golfer "takes the hazards" out of play! It may be wise to state that NO resonances shall occur in the range of 42-52% of "specified operating speeds." There are "pure whirl" conditions to worry about, and there are also subharmonic resonances at 1/2X, 1/3X, and 1/4X to worry about.

What is wrong with a first resonance at 57% of operating speed? Perhaps the operating company really does not need a 40% turn-down (reduction) of speed (it does allow a 30% reduction which is probably more realistic). They are probably going to operate at 105+% of maximum rates.

• One would not want to back away from unbalancing a rotor to (8 x Residual Unbalance Limit). This is a good test and proves the validity of the design work. Figure 1 shows a Steam Turbine Rotor unbalanced to 8 x 4W/N oz·in, yet the response is well below (< 0.6 mil pp) the balance acceptance level (Equation 1) of 1.2 mil pp.

Residual Unbalance Limit:

$$vibration,\ mil\ pp\ =\ \sqrt{\frac{12000}{speed}} \qquad (1)$$

• There are several ways to determine the Synchronous Amplification Factor (S.A.F.). API, because of multi-degree of freedom systems and concerns that the Bode plot could only clearly capture the top 30% of the resonance response during testing, chose to use the "Q Factor" definition that electrical engineers have used for *filter characteristics;* that is, *not only can filtered bandwidths be defined, but also the amplification factor and the shape factor at so many dB down (-) (for example, -20 dB). Therefore:*

$$S.A.F. = \frac{resonance\ frequency}{BW}; \qquad (2)$$

$BW = f_h - f_l$ @ 0.707 $Amplitude_{peak}$ $(-3dB)$ where f_h and f_l are the frequencies on either side of the peak at which the amplitude is 70.7% of the peak amplitude.

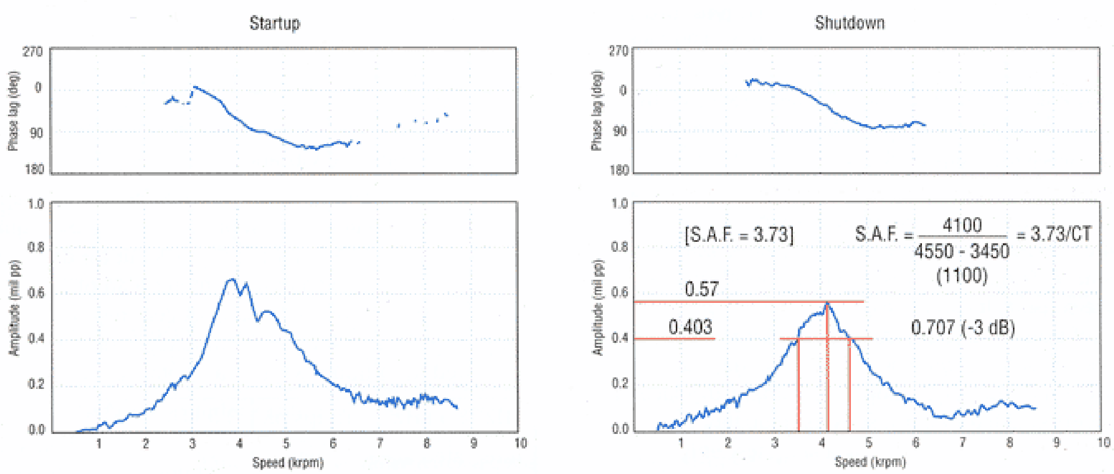

Figure 1. Steam turbine rotor unbalanced to 8 x U_{max} taken to 110% of speed. Nc @ 4100 cpm. S.A.F. = 3.73.

This S.A.F. is shown in Figure 1 for the turbine during the Bode plot at 8 x 4W/N oz · in. A residual unbalance of 4W/N oz · in (or 113.4W/N gm · inches) is equivalent to an ISO Grade of 0.665 mm/s. The W is the journal weight (or total rotor weight if not defined in two planes-left and right) in pounds. The N is the maximum continuous speed in rpm. $N_{mc} = 1.05 \times N_{design}$; $N_{trip} = 1.10 \times N_{mc}$ for turbines and centrifugal compressors (110% of max. Continuous Speed = 115.5% of Design Speed). I have seen large motors with an S.A.F. of 20, which is not good.

"If you haven't got the time or money to do it right, how come you have the time to do it over?"

• The Mechanical Running Test basically is a 4 hour endurance run at full speed against either a load or no load. Seal Leakage is normally tested for oil and other factors. Naturally, the rotor assembly integrity is under test, along with bearings, vibration, etc. Vibration sensors (shaft relative or seismic) and temperature sensors are per the latest API 670 specifications. I "never request" the contract vibration sensors be used on the test, but calibration OK'd.

• I have seen vibration, not related to the machine train in question, show up. I have seen many dynamometers with very poor vibration compared to the "driver" on test; that is, not related, but very bad. I have not been at a test where all the sensors were connected correctly. I want unfiltered scope displays or the equivalent on test at all times during the test. One must remember, again, that filters remove important information: 1X will "not" show "pure oil whirl." A 1X Bode plot will "not" show subsynchronous or supersynchronous (NOT 1X) resonance excitation. Bearing Pedestal resonance will probably not show on 1X plots. Polar plots show rotor bows better than Bode plots.

• Table 1 is not complete; that is, STD 546, Brushless Synchronous Motors, second edition, June 1997,which is not a part of the Subcommittee on Mechanical Equipment (S.O.M.E.), is not included. However, the table should give a "brush stroke" of the coverage over a large class of equipment.

API Standard 670 outlines the monitoring of equipment. It covers Displacement and Seismic Sensors, Temperature Sensors for Bearings, Axial Displacement Sensors at Thrust Bearings, and once-per-turn phase reference sensors. It can be used for the location and placement of sensors without monitoring. Some people have the sensor drillings provided (for future installation per API 670) and test in the shop; for example, displacement, eddy current, shaft relative sensors to qualify the machine's performance during a shop test only.

• It is important to understand that a Shop Test Value should be more stringent than the Operating Shutdown Value. One is doing a shop test on a NEW machine, hopefully in "mint" condition. That test should be good. Balance and vibration values always amaze me because if one would take the residual unbalance test level, for example, oz · in, and divide it by the rotor's weight in ounces, then the eccentricity, e, in inches, of the mass is determined. This will be a very low level, maybe 20 to 40 micro inches (0.000030 inches average) or 0.03 mil pk (= 0.06 mil pp). So where is the vibration coming from? Not from residual unbalance!

• There is another standard not listed, as it also does not cover a class of equipment, but affects all classes — API Standard 671, Special-Purpose Couplings. It is very important and defines the overhung mass of the coupling for balance. There are at least 3 minimum classes of balance that can be called out. Generically speaking, this would be (1) 4W/N oz · in, of the component balance with Mass being the overhung mass of the coupling and the random assembly residual unbalance limited to 40W/N; (2) Assembly Balance to a criteria of 4W/N oz · in, requiring match marking; (3) Component balancing to 4W/N oz · in followed by assembly balancing to 4W/N oz · in, also requiring match marking in assembly.

• There are Recommended Practices (RP) specifications and Publications (Publ) in the API specifications, and most are quite good and extensive. Publ 684 covers Rotor Dynamics. RP 686 covers Foundations, Grouting, Installation, Lube Systems, Commissioning, etc. RP 687 is a new one which will cover Rotor Repairs. It was started in 1995, with the 1st draft due to be in API S.O.M.E. review some time in 1998, and uses Centrifugal Compressors as the first template.

Some examples:

• Figure 1 shows a Steam Turbine Rotor, ~15,000 rated BHP, during a shop test wherein the rotor was unbalanced to 8 x the specified unbalance (4W/N oz · in or 113.4W/N gm · in). Vibration < 0.6 mil pp.

• The next example was a gearbox tested to the specifications in the latest edition of API 613 (the ISO equivalent of this specification (CD 13691) is out for draft vote in August 1997) using a dynamometer load of 1850 hp (greater than 10% of rated load). (A gear test without load is slightly worse than an "unloaded" motor test.) It was tested for 4 hours with an acceptance level of:

5 Hz- 2.5 kHz	0.15 ips pk	2.5 kHz-10 kHz	4 g pk
Integrated accelerometer & velocity sensor		Impedance matched accel. w/mounted resonance at 24 kHz.	

This gearbox never exceeded 0.1 ips pk nor 2 g pk at load and speed or lower. Furthermore, this gearbox, on commissioning in the field in May, 1996, ran at ~2.3-2.6 g pk @ 10.2 MW output for 1 week.

The subject gearbox was arranged for a "meshed-up" pinion. Often it is better to allow the pinion to rise in its bearing than to stay as a "down-meshed" pinion, by some specifications, because the low speed gear may actually lift at some percentage of full load and become unstable. This gear box was treated in this way and set on a concrete mezzanine block and grouted on sole plates. It was aligned in the field for the transient heat rise, which was ~40 mil on the pinion and ~30 mil on the gear. Since this is a generator drive from a steam turbine, the train will come to speed before the load (torque) is fully applied.

Most people don't understand that it is possible for a resonance to go through a speed ...as opposed to the speed going through a resonance—it's mind boggling.

When one uses displacement probes mounted orthogonally (90 degrees apart), the pinion eccentricity can be plotted during runup and load out. In Figure 2, a plot is made of the pinion shaft rising in its bearing clearance. Bearing designers are extremely interested in these type of plots. Bently Nevada's ADRE® for Windows/DAIU (Data Acquisition Interface Unit) allows one to do this shaft centerline plot (eccentricity plot). In the figure, the pinion rises up and right as speed increases (in complete agreement with the load vector plot submitted by OEM), then the load is applied. Voilà!

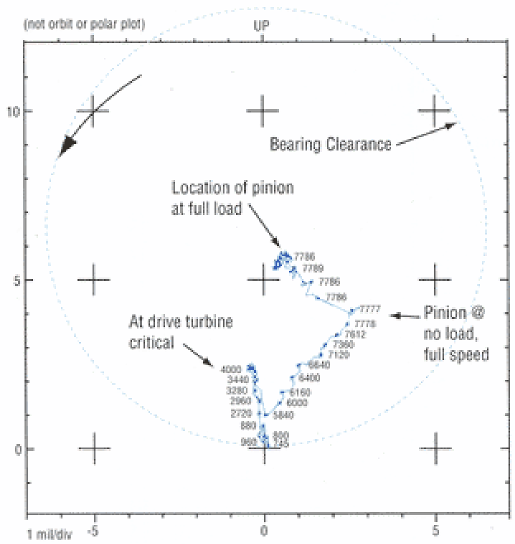

Figure 2. An eccentricity plot of a Pinion Shaft at the blind-end bearing. The pinion rises up and to the right as it comes to synchronizing speed (7778 rpm, generator at ~1800 rpm), passing through a drive turbine critical ~4000 rpm (bobby-pin loop). Then the load (torque) is applied and the pinion goes to near the center of the bearing clearance (circle equals the actual bearing clearance). While I seem to be the most interested in this, maybe others will be too.

• The next example is a centrifugal compressor during a Shop Test (the barrel compressor is later installed in a 1450 T/D Methanol syn gas train). The design speed is over 10,000 rpm with the trip speed ~12,000 rpm. The Bode and polar plots (Figure 3) show the startup during 4 hour running mechanical test.

The orbits (Figure 4) are elliptical on this rotor because it is a 5 tilt pad, load-on-pad (LOP) bearing which will have more stiffness vertically than horizontally; that is, non-symmetric (asymmetric) stiffness.

The half spectrum cascade plots (Figure 4) show only synchronous (1X) running speed components. This data is presented as part of the Test Package Data required, along with unfiltered orbits and timebase data, to assure that no significant, non-running speed (NOT 1X) vibration is present, such as an instability, rub, or misalignment.

Figure 5 is the Bode plot when the rotor has 3.685 gm · in of unbalance, with a maximum response <0.7 mil pp versus a test acceptance of ~1 mil pp.

• The final example shows the (full load, at speed) shop test of a centrifugal air compressor rotor, after repairs, in a vacuum bunker (Figure 6). This unit was balanced for the 2nd pivotal mode (approximately at operating speed). This example is shown to stress the importance of knowing "what to expect in the field." Is this rotor governed by API Testing Procedures? For the most part, NO! Should it be? YES! Has this rotor behaved badly in the field, with some added unbalance-rust, scale, water? Yes. Has it exceeded the 1/2 to 2/3 of bearing clearance criteria for fatigue damage to the bearings? Yes! [It was within 1/2 mil of the actual bearing clearance when it was shut down with excessive vibration (4 1/4 mil pp)]. Were the bearings damaged? NO! Could the rotor be turned by hand in the field? NO! Were the abradable labyrinth shaft seals in contact? YES!

Comments on the examples

Why could we operate many plants at 99.2% availability? Good designs, not installing past problems, and Shop Testing are a good start. Some comments to the data presented may help.

Figure 1 is from a steam turbine installed in a Petrochemical Plant - 3rd turbine generator at 10.2 MW, with a 42 MW GT Generator Set operating ~5 years, and a smaller, ~5 MW, steam turbine-gear-induction generator. Much of the customary extras were waived; for example, the need to validate the OEM's dynamics programs. The separation margin was specified, bearing designs were optimized, and the unbalance check at 8 x-allowed-unbalance was performed, along with many other control upgrades. This was the unbalanced run at 0.6 mil pp.

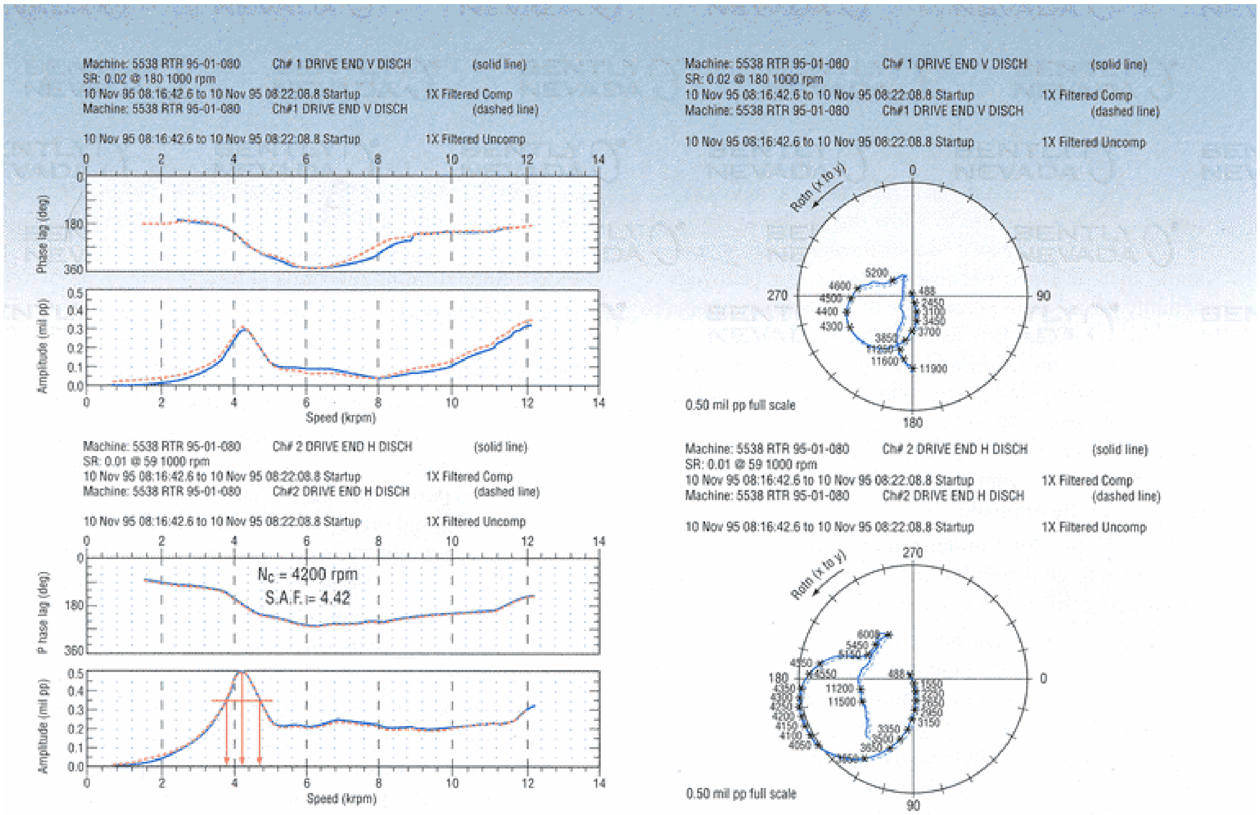

Figure 3. Bode and polar plots of a barrel compressor on 4 hour Mechanical Test (per API 617). The acceptance value of this test was ~1 mil pp. The Synchronous Amplification Factor was ~4.42. N_c pk 0.3 (vertical) and 0.5 (horizontal) mil pp.

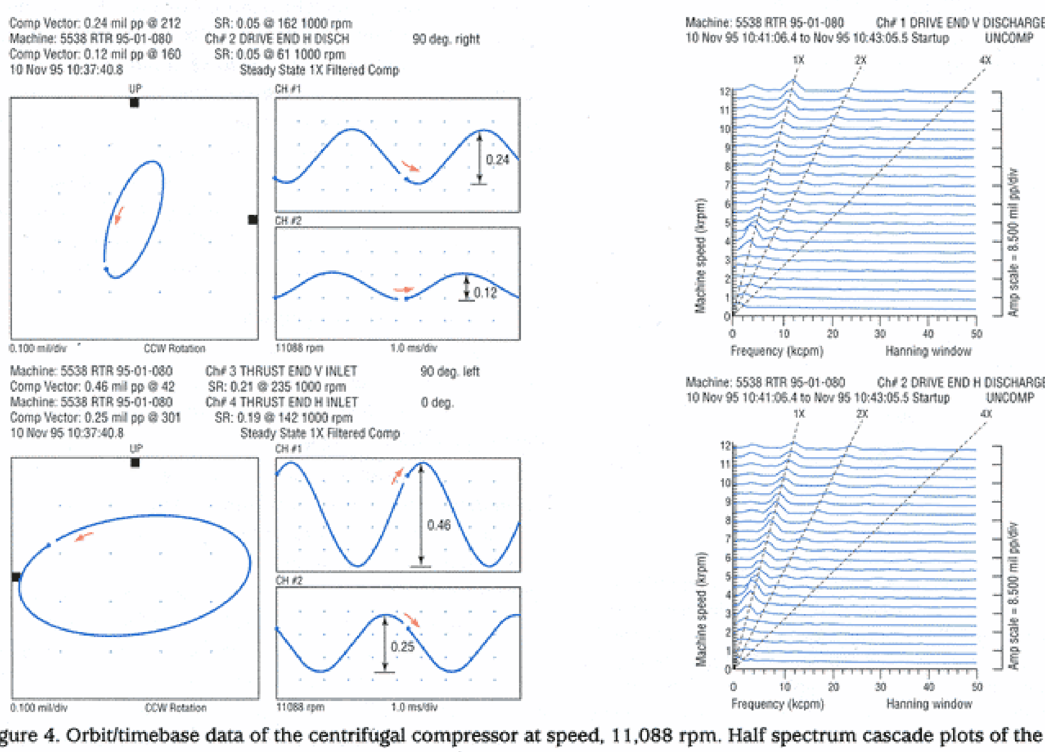

Figure 4. Orbit/timebase data of the centrifugal compressor at speed, 11,088 rpm. Half spectrum cascade plots of the compressor during runup at the end of the 4 hour endurance run. 1X would be the running speed.

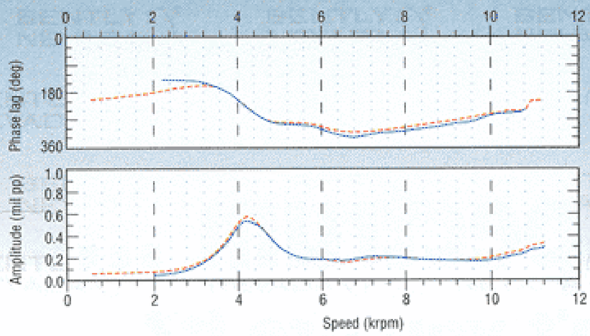

Figure 5. The Bode plot of the centrifugal compressor, after it has been unbalanced by 3.685 gm·in at the coupling. Test of the OEM rotor dynamics program; more importantly a test of damping under unbalance. The unbalanced vibration, ~0.7 mil pp, still passes the acceptance test for a balanced compressor.

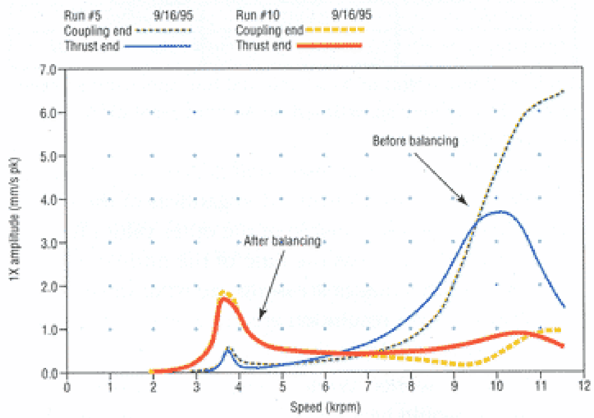

Figure 6. An air compressor rotor before and after balancing (at speed). It seems obvious that the second mode resonance exists near the operating speed. This resonance is balanced "at speed" (~10,200 rpm).

Example 2. This gearbox was specified to API Standard 613, 4th Ed., before it was issued. Can one do that? Sure. Everyone, who is anybody, has the draft copies. This was maybe the 1st gear tested to 4 g pk (AGMA is 10) & 0.15 ips pk. It passed at half that value, primarily because the specifications were good for quality for both the owner and OEM (the owner got reliability and the OEM got a good reputation).

Figure 2. This eccentricity plot was made because a couple of bearing improvements were added in this machine train, unfortunately not in the pinion and gear bearing. However, it does address one of the many different things about generator sets. They come to speed and then the load is applied. What does an up-meshed pinion do?

Figure 3. This was a highly-loaded syn gas centrifugal compressor typical of methanol and ammonia plants. A 4 hour mechanical test was performed. The installation time of a renewed large, ~1500 T/D plant depended on two of these compressors. This Bode and polar plot only confirm the smooth run in the shop. Vibration < 1/2 mil pp.

Figure 4 shows the timebase and orbit information for the same good running data. Furthermore, elliptical orbits are very common on 5 pad LOP bearings (nothing strange here). The major diameter of the orbit, per API 617, is the actual value of the test (similar to S_{max} in the ISO standards). It could be much greater than the vertical or horizontal amplitudes. It is another good check on phase accuracy, amplitude logic, forward precession, etc. The half spectrum cascade plot, to me, says there are no extraneous frequencies (NOT 1X) to be concerned about.

Figure 5 tries to be complete. A centrifugal compressor, operating at ~7877 rpm, was unbalanced at the coupling to show both static & pivotal mode response on the test stand.

Figure 6 simply confuses me in the technical world. Here is a compressor design that can be greatly improved by changing the main shaft, bearings, and seals without touching the impellers. It is operating at the second pivotal mode, which makes it very sensitive to any unbalance. Money is being spent in the wrong area.

Summary

This article was written to address the issue of shop testing. Is it right for you? It could easily be related to insurance for a good startup and successful long term operation. Unfortunately, in many cases, as the equipment gets into final assembly, the lack of project money (poorly planned or executed) often eliminates the last step...testing. It is an easy project decision, by the people holding the purse strings (controlling the money). In my opinion and experience, it is not a "wise" decision. I'm often reminded of the old maintenance proverb, "If you haven't got the time or money to do it right, how come you have the time to do it over?"

Has it been right for me? Very definitely. Yes! I was always a little embarrassed to be in the company of others who tested "more" than my company. People who do not do it suffer large, expensive losses.

References

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